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U1S S1847 S2011

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(54) Hydraulic damper

(57) A hydraulic damper comprises: a cylinder (13), a piston (14) slidable in the cylinder, communication passages (19) formed through the piston for communicating two liquid chambers (17, 18) defined in the cylinder with each other, and a damping force generation mechanism (24) effective when the damper is extended. The damping force generating mechanism (24) includes a valve disc (25) biassed by smaller diameter valve discs (26) or a spring (56) and inner and outer pressure chambers (28, 29). On extension the periphery of disc (25) deflects at one pressure, and discs (25, 26) deflect at a higher pressure.

A damping force is generated on damper contraction by fluid passing through passages (20) and past spring biassed valve disc (33). Similar valve arrangements are provided for allowing fluid flow between chamber (18) and an annular chamber surrounding cylinder (13) via a cylinder end closure.

Fig. 3

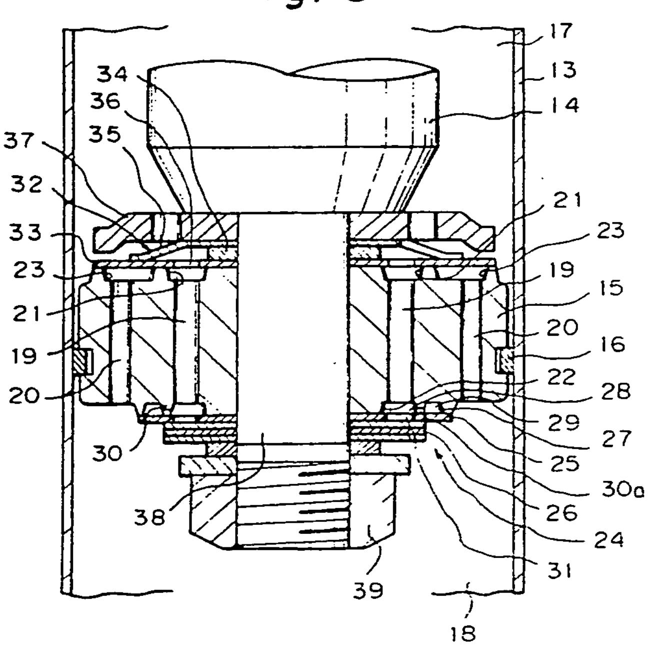

Fig. 13

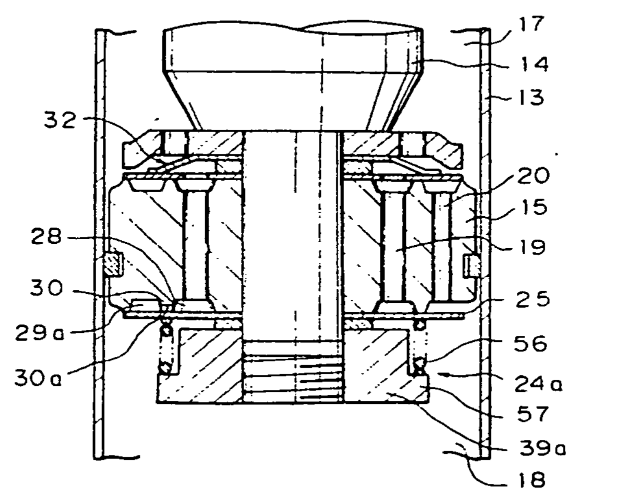

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Fig. 1

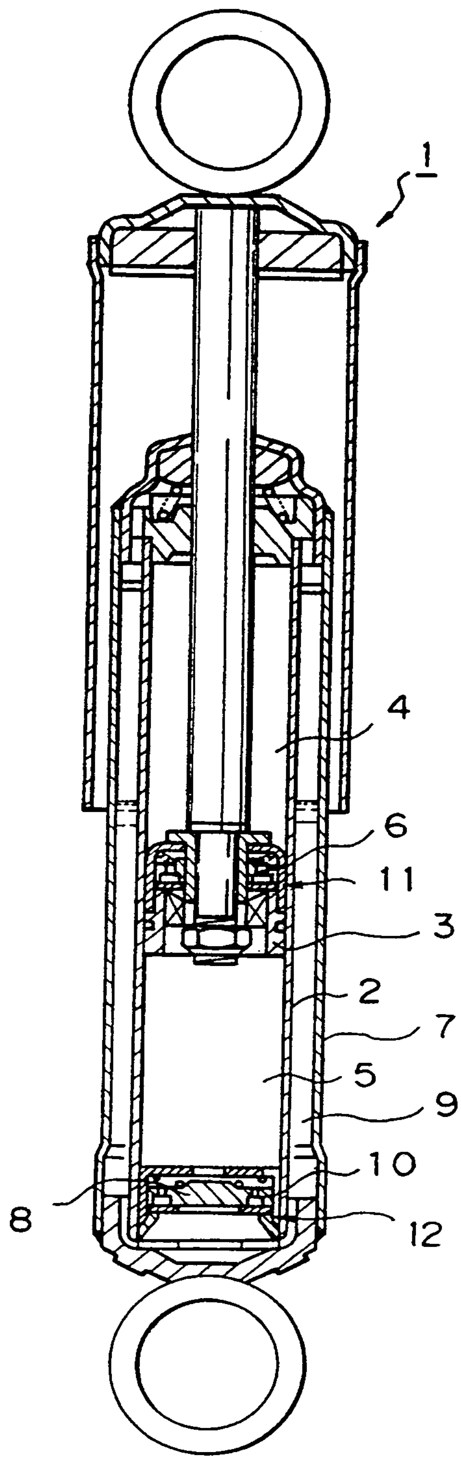


Fig. 2

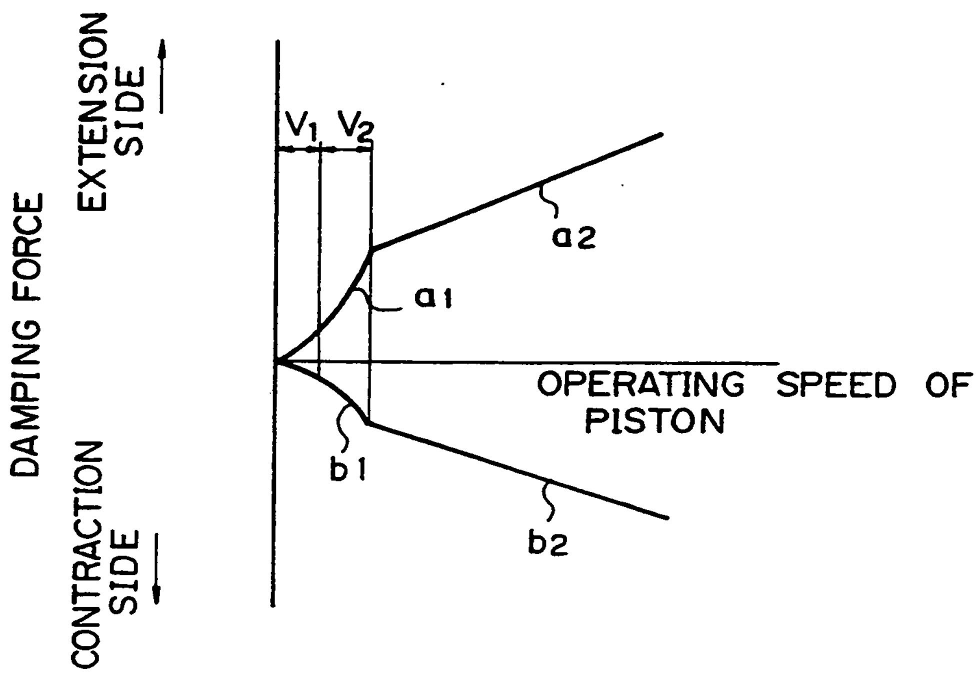

EXTENSION
SIDE
DAMPING FORCE
CONTRACTION
SIDE
V1
V2
a2
a1
b1
b2
OPERATING SPEED OF
PISTON

Fig. 4

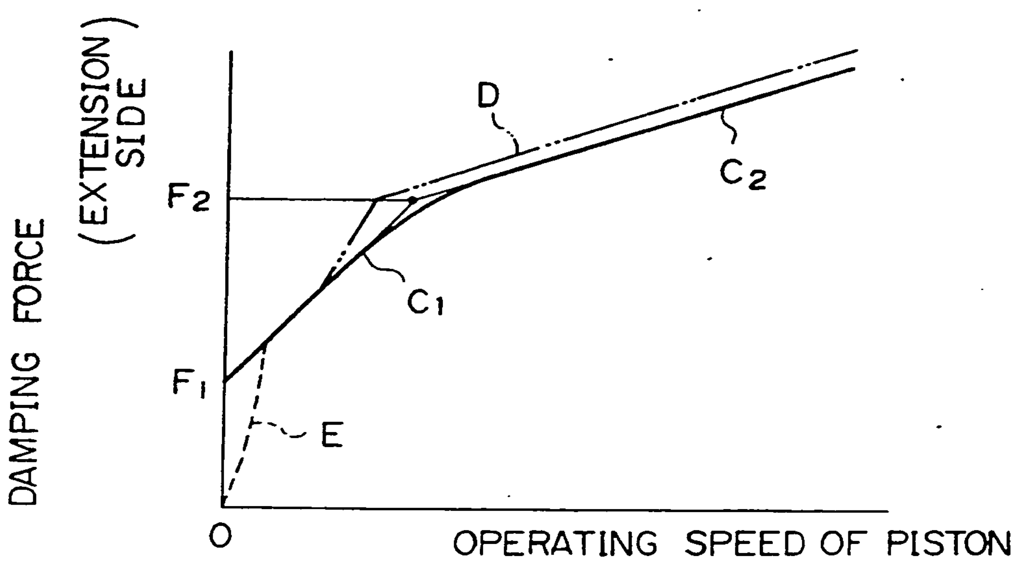

DAMPING FORCE
(EXTENSION
SIDE)
F2
F1
D
C2
C1
E
O
OPERATING SPEED OF PISTON

Fig. 3

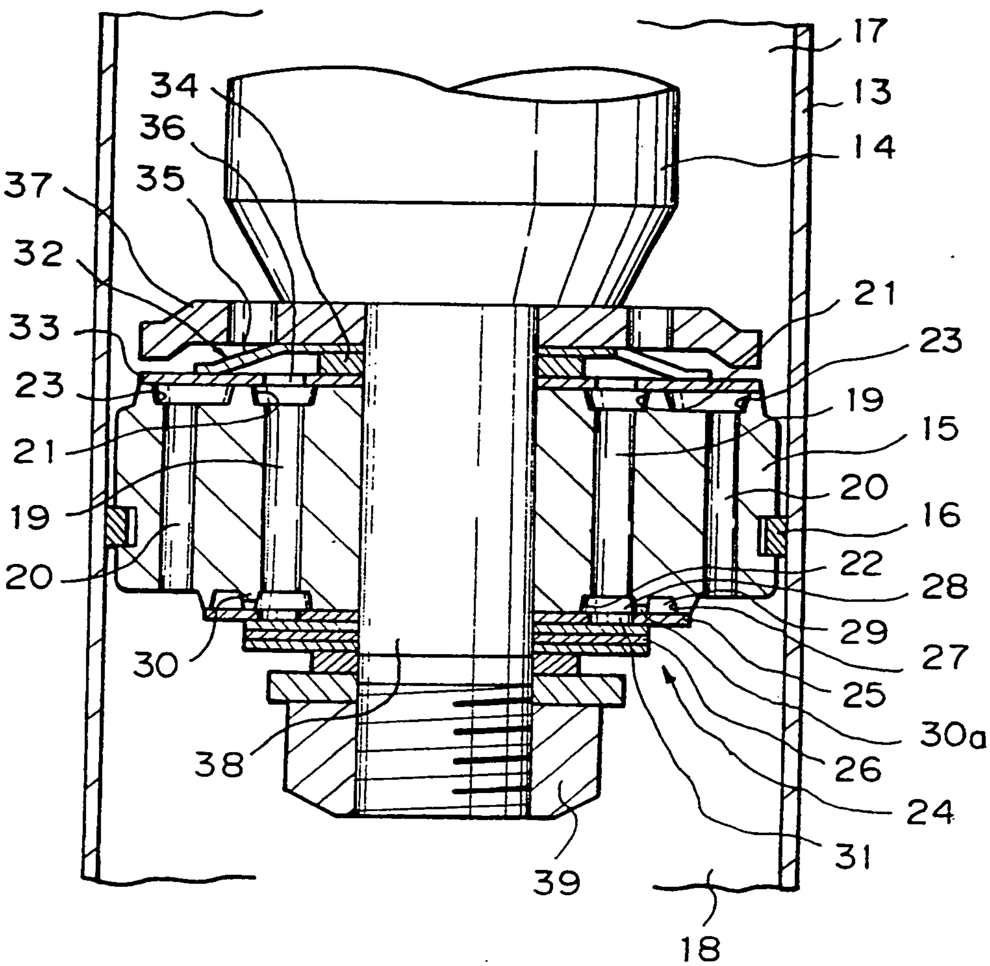

17
13
34
36
14
37
35
32
21
33
23
23
19
21
15
19
20
20
16
22
28
20
29
27
30
25
30a
38
26
24
31
39
18

Fig. 5

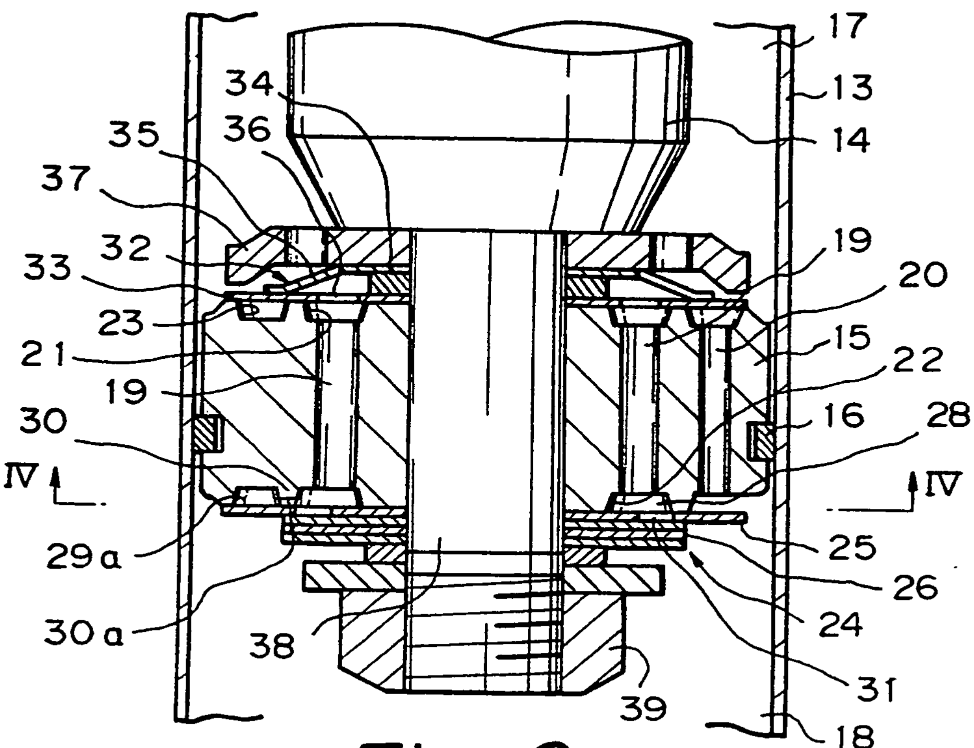

17
13
14
34
36
35
37
32
33
23
21
19
30
IV
29a
30a
38
39
18
31
24
26
25
IV
28
16
22
15
20
19

Fig. 6

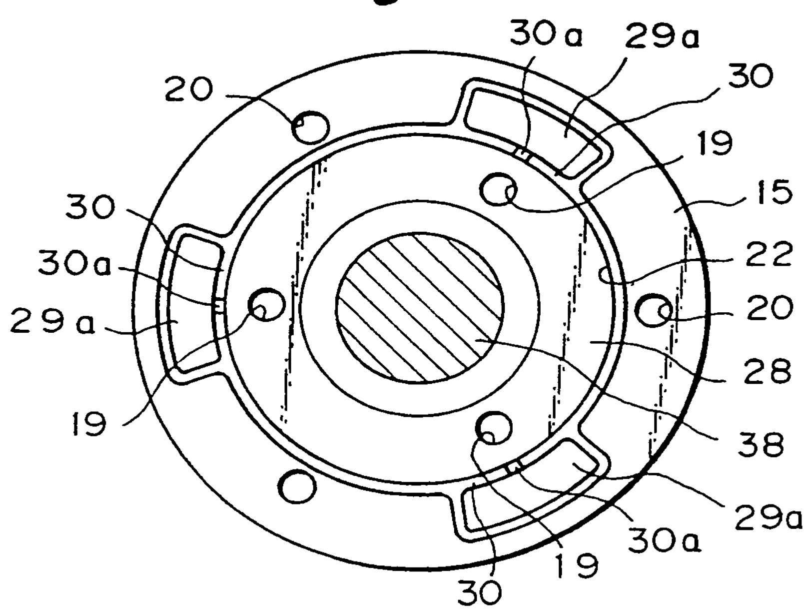

30a
29a
30
20
19
15
30
22
30a
20
29a
28
19
38
29a
30a
19
30

Fig. 7

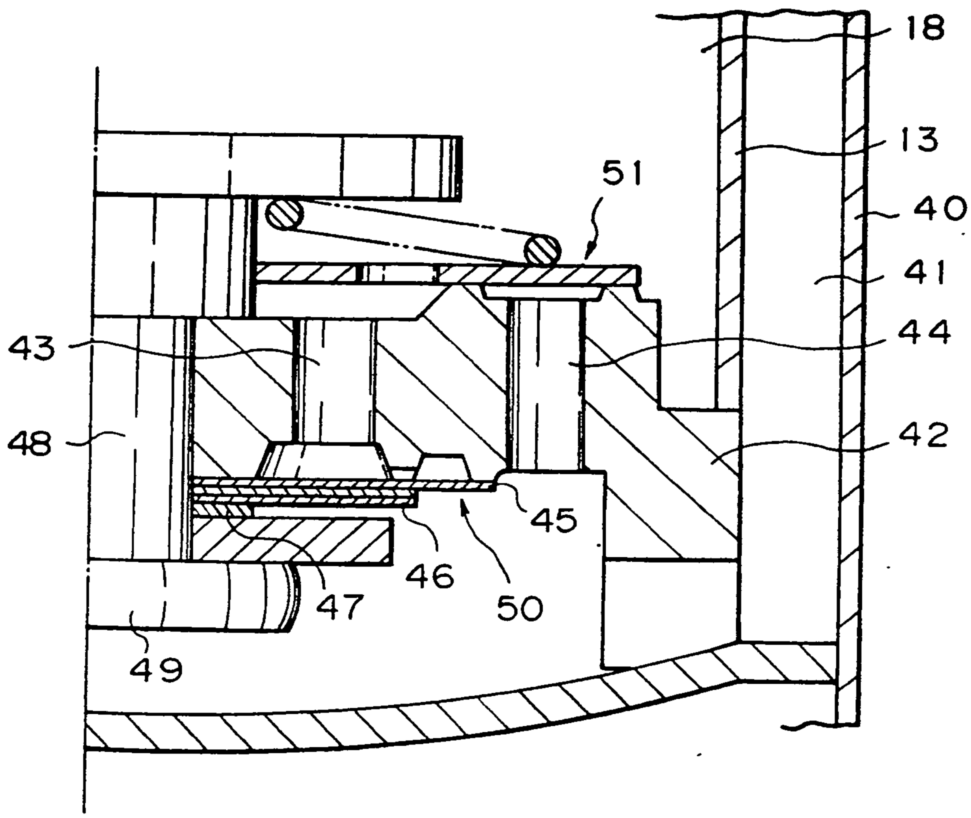


Fig. 8

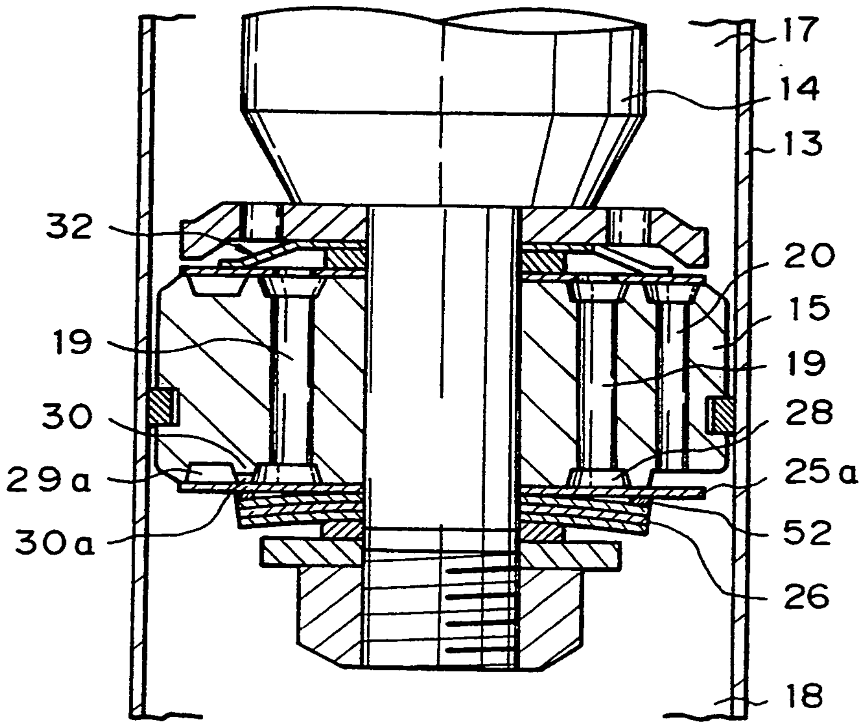

17
14
13
32
20
15
19
19
30
28
29a
25a
30a
52
26
18

Fig. 9

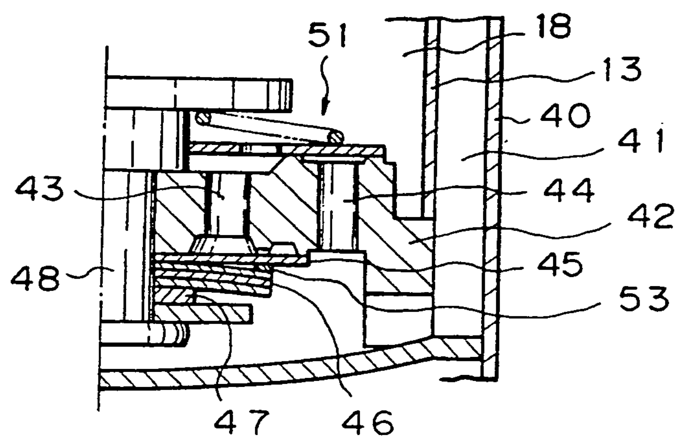

51
18
13
40
41
43
44
42
48
45
53
47
46

Fig. 10

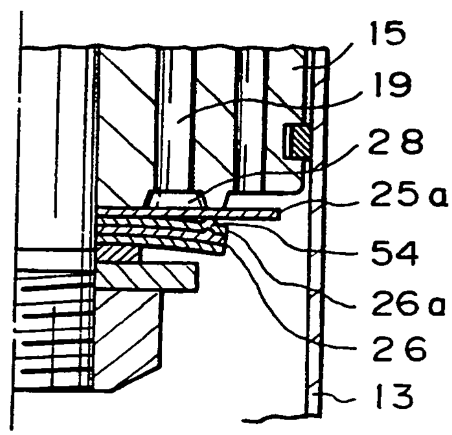


Fig. 11

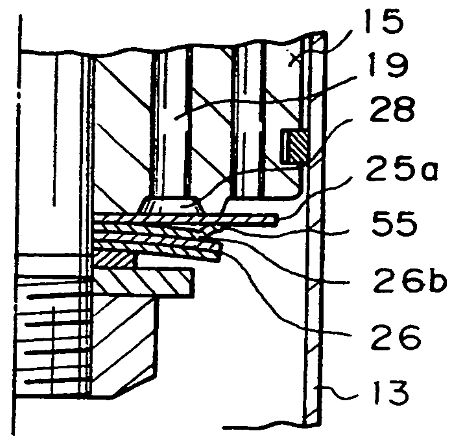


Fig. 12

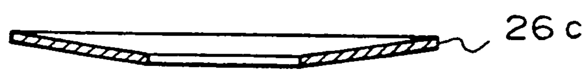


Fig. 13

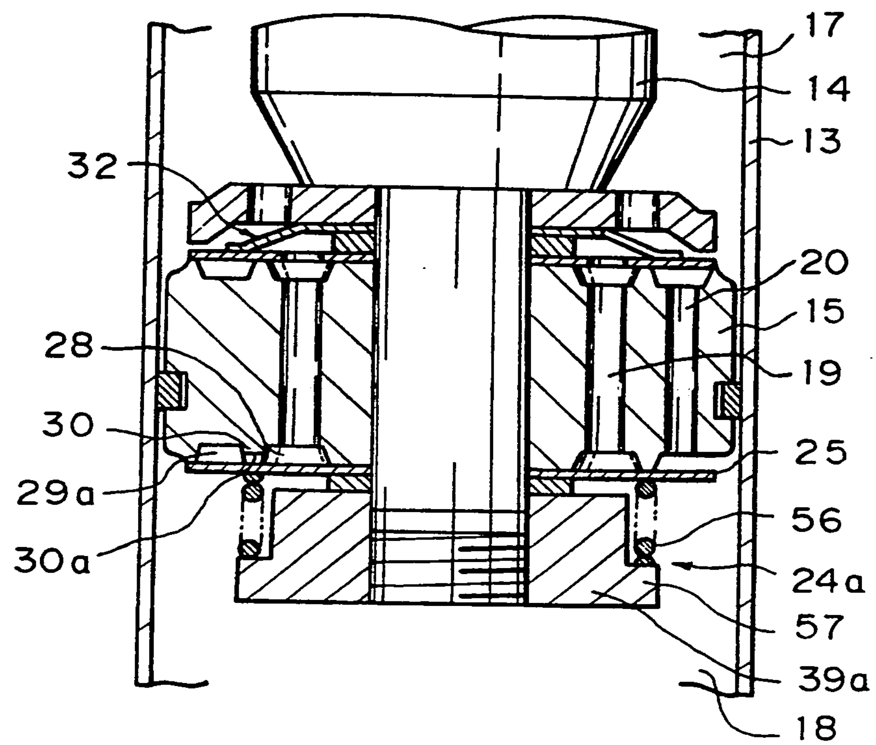

17
14
13
32
20
15
28
19
30
25
29a
56
30a
24a
57
39a
18

Fig. 14

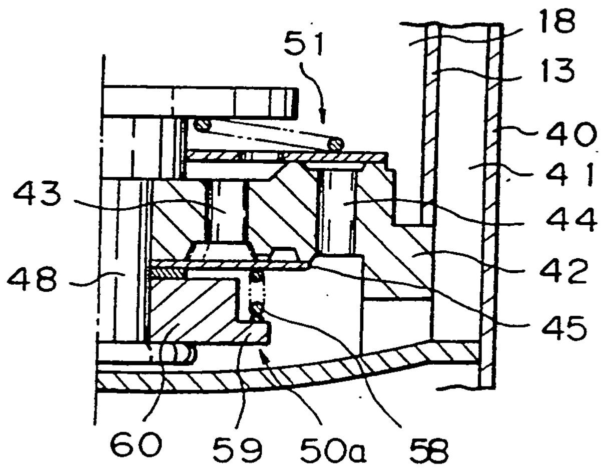

18
51
13
40
41
43
44
48
42
45
60
59
50a
58

Fig. 15

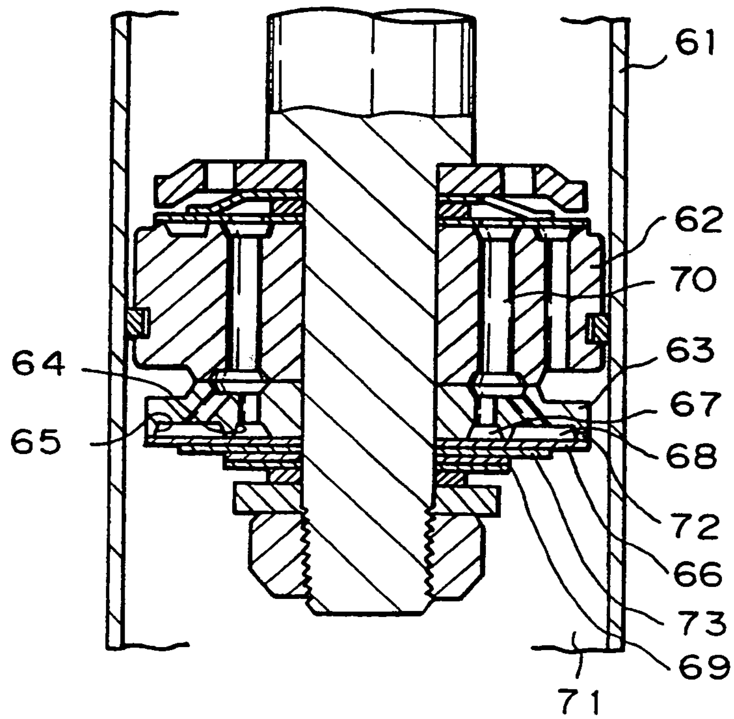


Fig. 16

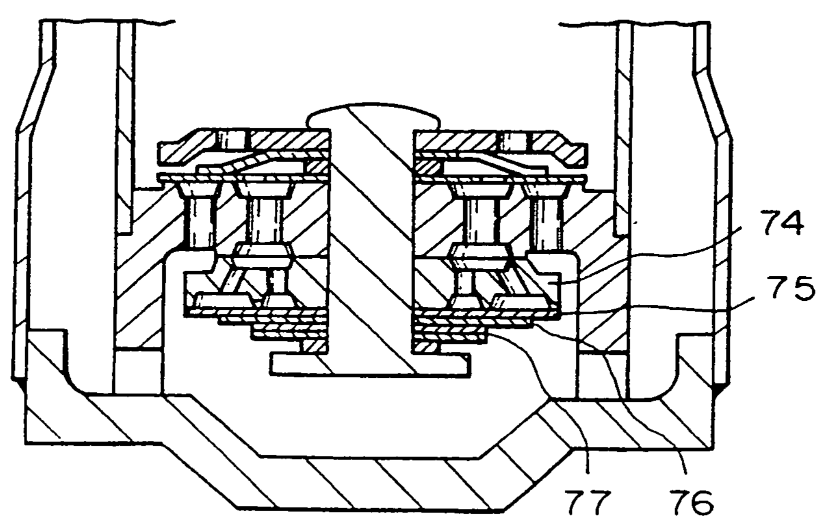


Fig. 17

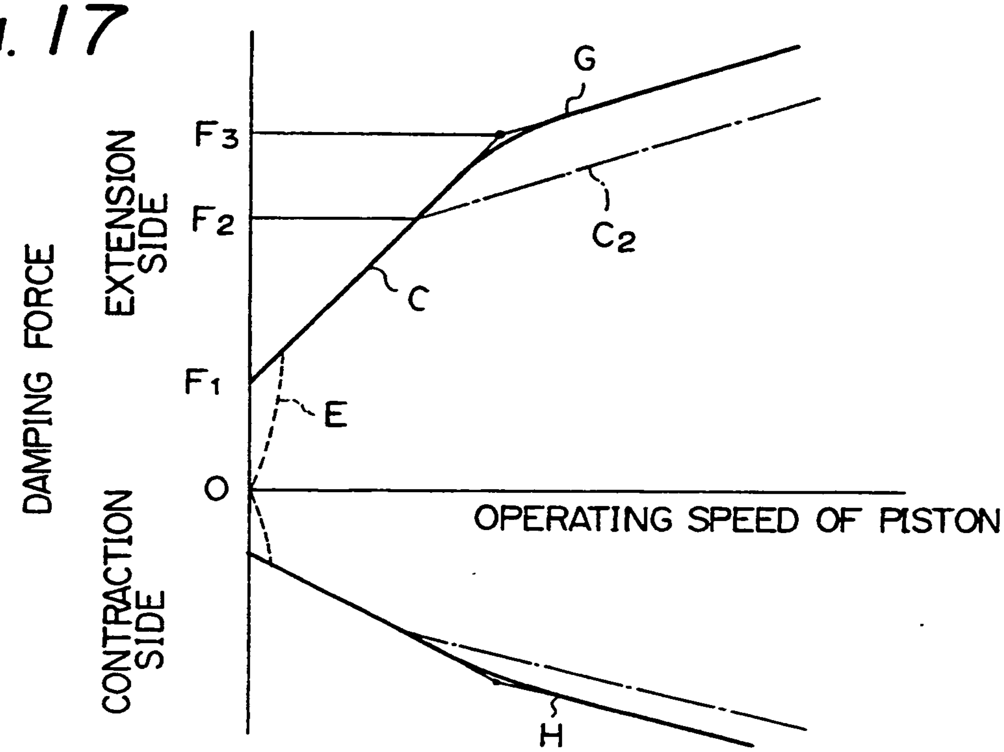
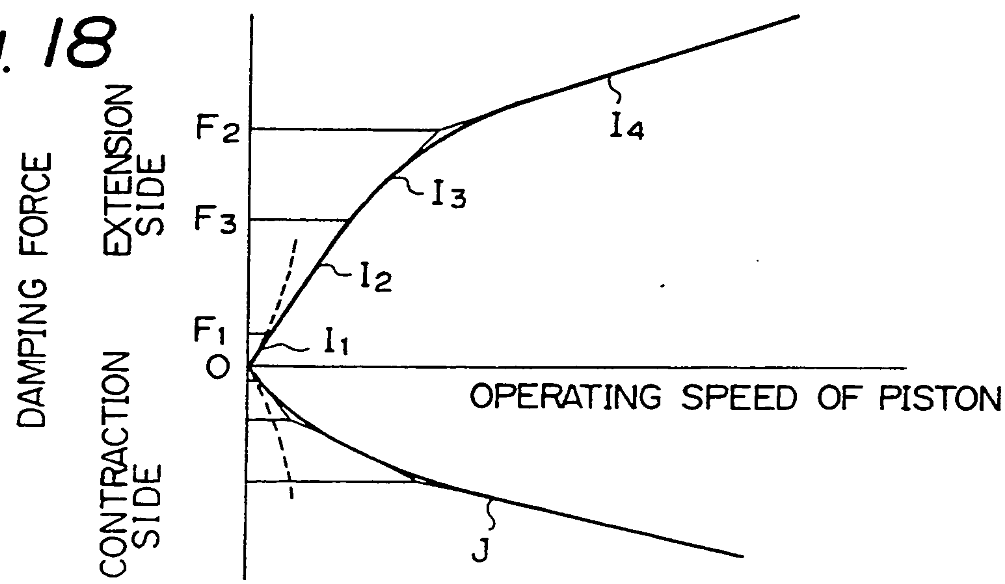

DAMPING FORCE
EXTENSION SIDE
CONTRACTION SIDE
G
F3
F2
C2
C
F1
E
O
OPERATING SPEED OF PISTON
H

Fig. 18


DAMPING FORCE
EXTENSION SIDE
CONTRACTION SIDE
F2
I4
I3
F3
I2
F1
I1
O
OPERATING SPEED OF PISTON
J

HYDRAULIC DAMPER

The present invention relates to a hydraulic damper adapted to use in a vehicle and the like.

As shown in Fig. 1, the hydraulic damper 1 according to a prior art which has been in general use is so constructed that the interior of a cylinder 2 is divided into two chambers 4, 5 by a piston 3 slidably fit in the cylinder 2, the chambers 4, 5 being respectively in communication with each other through a communication passage 6 formed in the piston 3, a chamber 9 defined between the cylinder 2 and an outer cylinderical body 7 and the chamber 5 in the cylinder 2 are in communication with each other at the bottom side of the hydraulic damper through a communication passage 10 formed through a partition member 8 mounted to the bottom part of the cylinder 2. In a hydraulic damper with such a construction, a sliding movement of the piston 3 in the cylinder 2 causes the flow movement of the hydraulic fluid therein through the respective communication passages 6, 11. At that time, the flow of the hydraulic fluid may be controlled by damping force generation mechanisms 11, 12 comprising a plurality of disc valves and orifice passages so that damping force may be generated.

The damping force characteristics as obtained by the hydraulic damper 1 of the above construction are as illustrated in Fig. 2. As it is seen from Fig. 2, when the operating speed of the piston 3 is slow, the hydraulic fluid is

throttled when it flows through the orifice passage, so that the characteristic thus generated is like that of a quadratic curve as designated by references a_1 and b_1, while, when the operating speed of the piston 3 is fast, the disc valve may be deflected due to hydraulic pressure, generating a linear characteristic as designated by references a_2 and b_2.

As explained above, the hydraulic damper 1 according to the prior art mentioned above provides the characteristic of a qudratic curve with regard to the characteristic caused by the orifice passage (or orifice characteristic), wherein at the initial operational stage of the piston (or the interval designated by V_1 in Fig. 2) a small damping force generated and the damping force will subsequently be rapidly increased.

It is to be noted, however, that a certain large damping force will be required at the initial stage of operation of the piston in order to prevent staggering of a vehicle during a normal driving to enhance stable driving thereby. However, the above-described orifice characteristic cannot adequately meet this requirement. Further, since a large damping force will have been generated during the period following the initial operational stage (or the period V_2 shown in Fig. 2), the vibration caused at the side of the wheels may be transmitted mincingly to the vehicle body whereby driving comfort may be impaired.

In this way, since the damping force characteristic provided by the hydraulic damper according to the prior art is not capable of adequately adapting the damping force to

the condition of a driving vehicle, most optimum driving stability and driving comfort cannot be attained according to the conventional hydraulic damper mentioned above.

The present invention has been proposed in view of the problems as pointed out above and has it as an object to provide a hydraulic damper which is capable of establishing an optimum damping force characteristic corresponding to specific vehicles.

The present invention provides a hydraulic damper comprising a cylinder, a piston slidably fit in the cylinder and partitioning the interior of said cylinder to two liquid chambers, the piston being provided with a communication passage formed therethrough for communicating the two liquid chambers with each other, and a damping force generation mechanism for generating damping force by controlling the flow of liquid through the communication passage caused by sliding movement of the piston in the cylinder, wherein the damping force generation mechanism includes a larger diameter disc valve disposed on the downstream side of the communication passage, a small diameter disc valve stacked on the larger diameter disc valve, the diameter of the smaller diameter disc valve being smaller than that of the larger diameter disc valve, an inner pressure chamber disposed radially inwardly on the opposite side of the larger diameter disc valve remote from the smaller diameter disc valve, the inner pressure chamber being in communication with the communication passage, and an outer pressure

chamber disposed on the same side as the inner pressure chamber radially outwardly of the inner pressure chamber, the outer pressure chamber being located radially outwardly of the outer circumference of the smaller diameter disc valve, the outer pressure chamber being in communicate with the communication passage.

In one embodiment, the damping force generation mechanism preferably includes a pre-set load setting means for giving the smaller diameter disc valve on initial deflection.

The present invention also provides a hydraulic damper comprising a cylinder, a piston slidably fit in the cylinder and partitioning the interior of the cylinder to two liquid chambers, the piston being provided with a communication passage formed therethrough for communicating the two liquid chambers with each other, and a damping force generation mechanism for generating damping force by controlling the flow of liquid through the communication passage caused by sliding movement of the piston in the cylinder, wherein the damping force generation mechanism includes a disc valve disposed on the downstream side of the communication passage, inner and outer pressure chambers disposed at different locations in the radial direction of the disc valve and respectively being in communication with the communication passage, the inner and outer pressure chamber being adapted to impose hydraulic pressure on the disc valve, and a biasing means disposed on the opposite side of the disc valve remote from the pressure chambers for biasing the disc valve toward the pressure chambers, the

biasing means being located in the location corresponding to that of a partition wall separating the inner and outer pressure chambers.

According to the present invention, the smaller diameter disc valve is stacked on the larger diameter disc valve, and the valves are designed such that they are entirely opened when the fluid pressure in the pressure chamber increases to a level which is higher than another level of hydraulic pressure at which the outer circumference portion of the larger diameter disc valve is deflected. Therefore, as the hydraulic pressure in the respective pressure chambers is increased due to the movement of the piston through the communication passage, the outer peripheral portion of the larger diameter disc value is firstly deflected so that the damping force in accordance with the valve characteristic will be generated. As the hydraulic pressure in the respective pressure chambers will further be increased, both the larger diameter disc valve and the smaller diameter disc valve are caused to be deflected so that the damping force in accordance with another valve characteristic will be generated. Thus, it is possible to provide a damping force characteristic with combined valve characteristics.

Since it is possible to change the pressure at which the respective disc valves are to be opened and the gradient of the valve characteristics by changing the rigidity of the respective disc valves. It is therefore possible to obtain a higher damping force at the initial stage of operation of

the piston in which the piston speed is slow, and to keep the damping force at a low level in the operational range in which the operating speed of the piston becomes higher. Thus, it is possible to obtain appropriate damping force characteristics corresponding to specific vehicles.

In a hydraulic damper further including a pre-set load setting means for giving the smaller diameter disc valve an initial deflection, it is also possible to make higher the pressure at which the smaller diameter disc valve is caused to open, without changing the rigidity of the discs and the number of the discs to be used, or even if the rigidity of the disc valve is reduced.

In the type of damper in which the disc valve is biased by a biasing member, as the hydraulic pressure in the respective pressure chambers is increased through the communication passages due to the movement of the piston, the outer circumference portion of the disc is first deflected to generate a damping force with a valve characteristic.

As the hydraulic pressure in the respective pressure chamber is further increased, the entire disc valve will be deflected against the biasing force of the spring member to generate a damping force of the different valve characteristics. In this manner, the damping force of the combined valve characteristics may be obtained.

Since it is possible to change the pressure level at which the disc valve is to open and the gradient of the valve characteristics by suitably changing the rigidity of the disc valve and the spring constant of the biasing means

or spring member, it is possible to establish at will appropriate damping force characteristics corresponding to the specific vehicles.

Fig. 1 is a vertical sectional view showing a entire hydraulic damper according to a prior art;

Fig. 2 is a diagram showing the damping force characteristics of the hydraulic damper according to a prior art shown in Fig. 1;

Fig. 3 is a vertical sectional view of the essential part of a hydraulic damper according to a first embodiment of the present invention;

Fig. 4 is a diagram showing the damping force characteristic of the hydraulic damper according to the first embodiment of the present invention;

Fig. 5 is a vertical sectional view of the essential part of a hydraulic damper according to a second embodiment of the present invention;

Fig. 6 is a sectional view taken along the line IV-IV in Fig. 5;

Fig. 7 is a vertical sectional view of the essential part of a hydraulic damper according to a third embodiment of the present invention;

Fig. 8 is a vertical sectional view of the essential part of a hydraulic damper according to a fourth embodiment of the present invention;

Fig. 9 is a vertical sectional view of the essential part of a hydraulic damper according to a fifth embodiment of the present invention;

Fig. 10 is a vertical sectional view of the essential part of a hydraulic damper according to a sixth embodiment of the present invention;

Fig. 11 is a vertical sectional view of the essential part of a hydraulic damper according to a seventh embodiment of the present invention;

Fig. 12 is a vertical sectional view of a smaller diameter disc valve to be employed in a hydraulic damper according to a eighth embodiment of the present invention;

Fig. 13 is a vertical sectional view of the essential part of a hydraulic damper according to a ninth embodiment of the present invention;

Fig. 14 is a vertical sectional view of the essential part of a hydraulic damper according to a tenth embodiment of the present invention;

Fig. 15 is a vertical sectional view of the essential part of a hydraulic damper according to a eleventh embodiment of the present invention;

Fig. 16 is a vertical sectional view of the essential part of a hydraulic damper according to a twelfth embodiment of the present invention;

Fig. 17 is a diagram showing the damping force characteristics obtained by the fourth embodiment shown in Fig. 8 and the fifth embodiment shown in Fig. 9; and

Fig. 18 is a diagram showing the damping force characteristics obtained by the eleventh embodiment shown in Fig. 15 and the twelfth embodiment shown in Fig. 16.

It is to be noted that the constructions shown in

Fig. 7, Fig. 9, Fig. 10, Fig. 11 and Fig. 14 are symmetrical in their left and right halves and only the right halves are shown in these drawings.

Preferred embodiments of the present invention will now be explained by referring to the accompanying drawings.

Fig. 3 illustrates a first embodiment of the present invention, the constitution of which will now be explained.

A piston 15 attached to a piston rod 14 is slidably fit in a cylinder 13 with a piston ring 16 therebetween. The interior of the cylinder 13 is divided into an upper cylinder chamber 17 and a lower cylinder chamber 18 by the piston 15. The piston 15 is provided with an extension and contraction side communication passages 19 and 20, respectively, formed therethrough and extending generally in parallel with the axis of the piston rod 14. These passages 19 and 20 respectively communicate the upper cylinder chamber 17 with the lower cylinder chamber 18. One end of the extension side communication passage 19 is opened to an annular groove 21 formed at the upper end surface of the piston 15 while the other end thereof is opened to an annular groove 22 formed at the lower end surface of the piston 15. On the other hand, one end of the contraction side communication passage 20 is opened to the lower cylinder chamber 18 and the other end of the passage 20 is opened to a groove chamber 23 formed in the upper end surface of the piston 15 radially outwardly of the annular groove 21.

An extension side damping force generation mechanisms

24 adapted to generate a dampng force in the extension stroke of the piston 15 is provided at the lower side of the piston, which side is the downstream side of the extension side communication passage 19. The constitution of this mechanism will next be explained.

A disc valve 25 of a larger diameter is seated on the lower end surface of the piston 15 and a plurality of disc valves 26 of a smaller diameter are stacked on the side of the larger diameter disc valve 25 remote from the extension side communication passage 19.

It is to be noted that the annular groove 22 which is formed in the lower end surface of the piston 15 and to which the extension side communication passage 19 opens is located radially inwardly of the outer circumference of the smaller diameter disc valve 26. It is also to be noted that an annular groove 27 is formed in the lower end surface of the piston 15 such that it is located radially outwardly of the annular groove 22, radially outwardly of the outer circumference of the smaller diameter disc valve 26 and radially inwardly of the outer circumference of the larger diameter disc valve 25. An inner pressure chamber 28 is defined by the annular groove 22 and the larger diameter disc valve 25 while an outer pressure chamber 29 is defined by the annular groove 27 and the larger diameter disc valve 25. The pressure chambers 28, 29 are communicated with each other through a plurality of throttling passages 30a formed in the partition wall 30 formed between said pressure chambers 28, 29. As explained later, the smaller diameter

disc valve is adapted to provide a fulcrum for the large diameter disc valve adjacent to a partitioning wall defined between the inner and outer pressure chambers when the outer circumference portion of the large diameter disc valve is deflected.

The rigidity of the smaller diameter disc valve 26 is so selected that the outer peripheral portion of the larger diameter disc valve 25 is first deflected away from the lower end surface of the piston 15 when the hydraulic pressure in the outer pressure chamber 29 increases to F_1 and then the smaller diameter disc valve 26 is caused to open when the hydraulic pressure in the inner pressure chamber increases to F_2 which is larger than F_1. It is to be understood that the rigidity of the disc valves 25, 26 may be determined by selecting the wall thickness and the material thereof. Alternatively, the large diameter disc valve 26 may be provided with a plurality of holes 31 spaced apart from each other in the circumferential direction thereof as shown in Fig. 3 so as to reduce the rigidity of the outer peripheral portion of the larger diameter disc valve 25. Those holes are preferably formed in an area of the large diameter disc valve which is overlaid with the smaller diameter disc valve. The holes are also located to be adjacent the outer circumference of the smaller diameter disc valve.

A check valve mechanism 32 is provided on the upper end surface of the piston 15. The check valve mechanism will now be explained.

A disc valve 33 is seated on the upper end surface of the piston 15 in such a manner as to cover the annular grooves 21, 23 formed on the upper end surface. The disc valve 33 is biased against the piston 15 by a leaf spring 35 attached to the piston 15 through a retainer 34. A plurality of bores 36 are formed through the disc valve 33 at the locations corresponding to the annular groove 21 to which the extension side communication passage 19 opens, and the bores 36 serve to communicate the upper cylinder chamber 17 with the extension side communication passage 19. It is to be noted that numeral 37 in Fig. 3 designates a washer adapted to restrict deflection of the disc valve 33.

In the check valve mechanism 32 constructed in this way, as the hydraulic pressure in the contraction side communication passage 20 is increased in the contraction stroke of the piston, the disc valve 33 is caused to deflect against the biasing force of the leaf spring 35 so that the hydraulic fluid is caused to flow from the lower cylinder chamber 18 to the upper cylinder chamber 17.

It is to be understood that the damping force in the contraction strokes will be generated by a contraction side damping force generation mechanism provided on the bottom side of the hydraulic damper, which is not shown.

The check valve mechanism 32, the piston 15 and the damping force generation mechanism 24 are fitted on a smaller diameter portion 38 of the piston rod 14 and tightened and secured to the smaller diameter portion by a nut 39 threaded on the end of the portion 38.

The operation of the hydraulic damper constituted as above explained will now be explained with reference to Fig. 3 and Fig. 4.

First, as the piston 15 is pulled upwardly by way of the piston rod 14 in the extension stroke, the pressure in the upper cylinder chamber 17 increases, causing the hydraulic fluid to flow through the extension side communication passage 19.

At this time, the hydraulic fluid flaws into the inner pressure chamber 28 from the extension side communication passage 19 and further flows through the throttling passage 30a to the outer pressure chamber 29, increasing thereby the hydraulic pressure in the respective pressure chambers 28, 29. When the hydraulic pressure in the outer pressure chamber 29 is increased to the level of F_1, the outer peripheral portion of the larger diameter disc valve 25 is deflected to open the valve, generating the damping force as represented by the line C_1 in Fig. 4. As the operating speed of the piston is further increased, the flow of the hydraulic fluid from the inner pressure chamber 28 to the outer pressure chamber 29 is restricted by the throttling passage 30a and, when the hydraulic pressure in the inner pressure chamber 28 is increased to the level of F_2, the larger diameter disc valve 25 and the smaller diameter disc valve 26 are entirely deflected to open, generating the damping force as represented by the line C_2 in Fig. 4.

The valves of pressure such as F_1 and F_2 and the gradient of the lines C_1 and C_2 in the damping force

characteristic curve as shown in Fig. 4 may be easily adjusted by changing the amount of initial deflection and the rigidity of the respective disc valves 25, 26.

Furthermore, if the opening area of the throttling passage 30a communicating the pressure chambers 28, 29 with each other is changed to be smaller, the orifice characteristic may be added to the valve characteristic explained above, so that such a damping force characteristic as shown by the line D (two dot chain line) in Fig. 4 may be attained.

It is also possible to attain such a damping force characteristic as shown by the line E (dashed line) in Fig. 4 in which an orifice characteristic is added, by forming a cut-out (not shown) at the outer circumference portion of the larger diameter disc valve 25 to define an orifice passage for communicating the communication passage 19 with the lower cylinder chamber 13.

In this manner, the damping force to be generated may be set at a high level even in such a range as the operating speed of the piston 15 is slow by executing various modes of adjustment and, since optimum damping force characteristics corresponding to specific vehicles may be established, excellent driving reliability and driving comfort may be provided.

Another embodiment of the present invention will be next explained.

The second embodiment shown in Fig. 5 and Fig. 6 is different from the first embodiment only in the configuration of an outer pressure chamber 29 formed in the lower end

surface of a piston 15. More specifically, three outer pressure chambers 29a are provided at the equal distance from the axis of the piston 15 and equispaced from each other in the circumferential direction. Contraction side communicatin passages 20 are formed at the portions between the respective adjacent two pressure chambers 29a. Since the remaining construction is the same as that of the first embodiment, same numerals are denoted to designate the same elements and the explanation thereof is not repeated. Since the operation is also the same as that of the first embodiment, the explanation thereof is not repeated.

In the third embodiment as shown in Fig. 7, the present invention is applied to the bottom side of the hydraulic damper. It is to be noted in connection with this embodiment that the damping force in the contraction stroke is generated at the bottom side of the hydraulic damper.

A partition member 42 adapted to separate a lower cylinder chamber 18 and an auxiliary chamber 41 defined between a cylinder 13 and an outer cylindrical body 40 is formed with a contraction side communication passage 43 which communicates the lower cylinder chamber 18 with the auxiliary chamber 41 and an extension side communication passage 44. On the downstream side of the contraction side communication passage 43, there are stacked along a shaft member 48 attached to the partition member 42, a larger diameter disc valve 45, a plurality of smaller diameter disc valves 46 and a spacer 47 in this order. These parts are jointed together by a nut 49, thereby constituting a contraction side damping force generation mechanism 50.

On the downstream side of the extension side communication passage 44, there is provided a check valve mechanism 51 which allows the hydraulic fluid in the auxiliary chamber 41 to flow into the lower cylinder chamber 18 through the communication passage 44 only in the course of the extension stroke of the piston.

Since the operation of this embodiment is also the same as that of the first embodiment already described, the explanation thereof is not repeated.

Figs. 8 to 12 show other embodiments in which a pre-set load setting means is provided for giving an initial deflection to the smaller diameter disc valves 26 and 47 in the second and third embodiments, respectively. The same members as those of the aforementioned embodiments are denoted by the same numerals and the detailed explanation thereof is not repeated.

The difference between the fourth embodiment shown in Fig. 8 and the second embodiment above described is in that a pre-set load setting means is provided for giving a smaller diameter disc valve 26 with an initial deflection. The pre-set load setting means includes a ring 52 interposed between the outer peripheral end portion of the smaller diameter disc valve 26 and a larger diameter disc valve 25a. It is to be noted that no holes 31 for reducing the rigidity of the larger diameter disc valve 25a is formed in this embodiment.

According to this embodiment, since the outer peripheral portion of the smaller diameter disc valve 26 is

deflected by the ring 52 away from the larger diameter disc valve 25a, so that the pressure F_3 at which the smaller diameter disc valve 26 is to open may be changed to a level higher than F_2 according to the second embodiment as shown in Fig. 17 (this damping force characteristic is represented by the line G in Fig. 17). The pressure F_3 may be appropriately adjusted by changing the thickness (the width in the axial direction) of the ring 52 to alter the initial deflection for the smaller diameter disc valve 26.

In order to change the pressure at which the smaller diameter disc valve is to open to a higher level, the thickness of the smaller diameter disc valve 26 may be thicker so as to increase the rigidity thereof. Alternatively, the number of discs to be used may be increased. However, these methods of adjustment will necessarily cause the proportion of increase of the damping force (the gradient of the line G in Fig. 17) to be so large that a desired damping force characteristic may not be attained. Further, the weight of the disc valve will be increased and the entire length of the hydraulic damper will be longer. According to the present embodiment on the contrary, the pressure at which the smaller diameter disc valve 26 is to open may be changed to a high level without changing the proportion of increase of the damping force and without causing the weight and the length of the hydraulic damper to be increased. Thus, the hydraulic damper may be made light and compact.

The fifth embodiment shown in Fig. 9 is characterized in that a pre-set load setting means is provided for giving

an initial deflection to the smaller diameter disc valve 46 employed in the third embodiment. The pre-set load setting means in this embodiment includes a ring 53 interposed between the outer peripheral end portion of the smaller diameter disc valve 46 and a larger diameter disc valve 45. Since the operation of this embodiment is the same as that of the fourth embodiment as described above, the explanation thereof is omitted here (the damping force characteristic attained by this embodiment is represented by the line H in Fig. 17).

The sixth embodiment shown in Fig. 10 is characterized in that a smaller diameter disc valve 26 is provided with a projection 54 formed integrally therewith adjacent to the outer peripheral portion thereof. This projection functions as a pre-set load setting means. The tip end of the projection 54 is so designed as to abut against a larger diameter disc valve 25a to provide the smaller diameter disc valve 26 with an initial deflection. It is to be understood that this projection 54 may be formed partially or entirely along the circumference of the smaller diameter disc valve 26a. It is also to be noted that the smaller diameter disc valve 26a formed with the projection 54 may be turned upside down so that the tip end of the projection 54 will be in abutment with another small diameter disc valve 26.

The seventh embodiment as illustrated in Fig. 11 is characterized in that a smaller diameter disc valve 26b with a stepped portion 55 formed by bending is used in place of the smaller diameter disc valve 26a formed with the

projection 54 in the sixth embodiment. This stepped portion 55 is so designed as to abut against a larger diameter disc valve 25a, such that the smaller diameter disc valve 26b may be provided with an initial deflection. It is also to be noted that this stepped portion 55 may be formed partially or entirely along the circumference of the smaller diameter disc valve 26b. The smaller diameter disc valve 26b formed with said stepped portion 55 may be turned upside down so that the stepped portion 55 may be in abutment with another smaller diameter disc valve 26. The function of the sixth and seventh embodiments is similar to that of the fourth embodiment, but the costs for the former two embodiments may be reduced by the cost of the ring 52 which is used in the fourth embodiment. It is also possible to apply the smaller diameter disc valves 26a, 26b of the construction used in the sixth and seventh embodiments to the fifth embodiment so as to provide the smaller diameter disc valve 46 with an initial deflection in stead of using the ring 53 for this purpose.

The eighth embodiment illustrated in Fig. 12 represents another pre-set load setting means. This means comprises a disc valve 26c of a smaller diameter which is deformed into a dish-shape in advance. By assembling this smaller diameter disc valve 26c to be flat (or in the manner similar to the smaller diameter disc valves 26, 46 in the first and the third embodiments), the smaller diameter disc valves 26 are provided with initial deflection. It is to be noted in this case that the pressure at which the smaller

diameter disc valve 26c is to open may be suitably adjusted by changing the amount of deformation of the disc valve into a dish-shape or the number of the dish-shaped disc valves to be used.

A ninth embodiment of the present invention will now be explained by referring to Fig. 13.

According to this embodiment, a spring 56 is used in place of the smaller diameter disc valve 26 in the second embodiment. The same members as those used in the second embodiment are denoted with the same numerals in this embodiment and detailed explanations thereof are not repeated here.

A flange portion 57 is formed integrally with a nut 39a which is used to assemble an extension side damping force generation mechanism 24a. A spring member 56 comprising a coil spring is interposed between the flange portion 57 and a larger diameter disc valve 25. It is to be noted that the position at which the spring member 56 abuts against the larger diameter disc valve 25 corresponds to the location of a partition wall 30 between an inner pressure chamber 28 and an outer pressure chamber 29a.

In case that the spring member 56 is interposed in such a manner as to abut against the larger diameter disc valve without biasing it, it will perform the same function as that of the smaller diameter disc valve 26 in the second embodiment. In case that the spring member is interposed in such a manner as to bias the larger diameter disc valve 25, it will perform an identical function to that of the smaller

diameter disc valve 26 in the fourth embodiment in which an initial deflection is given by the pre-set load setting means. It is also possible to attain various damping force characteristics by changing the rigidity of the larger diameter disc valve and the elasticity of the spring member 56.

The tenth embodiment illustrated in Fig. 14 has applied a spring member 58 to a contraction side damping force generation mechanism 50a provided at the bottom side of the hydraulic damper in place of the smaller diameter disc valve 46 in the third embodiment, in a similar manner to that of the ninth embodiment.

According to this embodiment, a retainer 60 integrally formed with a flange 59 is provided and a spring member 58 comprising a coil spring is interposed between the flange portion 59 and a larger diameter disc valve 45. It is to be understood that the function of this embodiment is the same as that of the ninth embodiment as described above.

Fig. 15 shows an eleventh embodiment which enables the damping force characteristics to be varied smoothly, the constitution of this embodiment will now be described.

A retainer 63 is connected to the lower end surface of a piston 62 slidably fit within a cylinder 61, and two concentric annular grooves 64, 65 formed in said retainer 63 and a larger diameter disc valve 66 which abuts against the retainer 63 define an inner pressure chamber 67 and an outer pressure chamber 68. The inner pressure chamber 67 is disposed radially outwardly of the outer circumference of a smaller diameter disc valve 69. On the other hand, the

outer pressure chamber 68 is disposed radially outwardly of the outer circumference of the smaller diameter disc valve 69 and radially inwardly of the outer circumference of the larger diameter disc valve 66. An extension side communication passage 70 is branched at the jointed portion of the piston 62 and the retainer 63 to communicate with the inner pressure chamber 67 and the outer pressure chamber 68.

At the outer circumference of the retainer 63 in abutment with the larger diameter disc valve 66, there is formed an orifice passage 72 comprising a cut-out which communicates the outer pressure chamber 68 with a lower cylinder chamber 71.

Between the larger diameter disc valve 66 and the smaller diameter disc valve 69, there is interposed an intermediate diameter disc valve 73 which has the diameter smaller than that of the larger diameter disc valve 66 and larger than that of the smaller diameter disc valve 69. The rigidity of the intermediate diameter disc valve 73 is so selected that the outer periphery portion of the larger diameter disc valve 66 is deflected when the pressure in the outer pressure chamber increases to F_1 and the intermediate diameter disc valve 73 is opened when the pressure in the outer pressure chamber increased to F_3 which is larger than F_1 ($F_1 < F_3$). In the meantime, the rigidity of the smaller diameter disc valve 69 is so selected that it is opened when the pressure in the inner pressure 67 increases to F_2 which is larger than F_3 ($F_2 > F_3 > F_1$).

The remaining constitution in Fig. 15 is the same as

that of the first embodiment, so the explanation thereof is not repeated.

The operation of the embodiment having the constitution as above described will next be explained by referring to Fig. 15 and Fig. 18.

As the piston 62 is pulled upwardly in the course of extension stroke and the hydraulic fluid in the communication passage 70 is caused to move, thereby increasing the pressure in the inner pressure chamber 67 and the outer pressure chamber 68, the damping force of orifice characteristic as represented by the line I_1 in Fig. 18 will firstly be generated by the orifice passage 72, and then, as the hydraulic pressure in the outer pressure chamber 68 will be raised to the level of F_1, the outer peripheral portion of the larger diameter disc valve 66 will be deflected to generate the damping force of the characteristic represented by the line I_2 in Fig. 18. When the pressure in the outer pressure chamber 68 is increased to the level of F_3, then the intermediate diameter disc valve 73 will be deflected to generate the damping force represented by the line I_3 in Fig. 18. When the hydraulic pressure in the inner pressure chamber 27 is further raised to the level of F_2, the smaller diameter disc valve 69 is deflected to generate the damping force represented by the line I_4 in Fig. 18.

In this way, by providing an optimum combination of the damping force of orifice characteristic and the damping force provided successively by deflection of the respective disc valves 66, 69, 73, smooth damping force characteristics may be attained.

Although in the embodiment as described above, three kinds of disc valves, or the larger diameter disc valve 66, the intermediate diameter disc vale 73 and the smaller diameter disc valve 69 have been employed, more than four kinds of disc valves having different diameters, the smaller diameter disc valves providing higher pressure at which the concerned valves are to open, may be employed. The more disc valves having different diameters are used, the smooth will be the damping force characteristics.

The twelfth embodiment shown in Fig. 16 has applied the constitution of the eleventh embodiment to a damping force generation mechanism at the bottom side of the hydraulic damper. Since the constitution of the retainer 74 and the constitutions of the respective disc valves 75, 76 77 shown in Fig. 16 are identical to that of the eleventh embodiment and the constitution of the remaining elements is identical to that of the third embodiment, their explanations are omitted here. It is also to be noted that the function is the same as that of Fig. 11 (the damping force characteristic is represented by the line J in Fig. 18).

It is apparent that the present invention should not be limited to the respective embodiments described above, and it may be constituted in the following manner as well.

With regard to the inner pressure chamber and the outer pressure chamber in the respective embodiments, any configuration may be applied as long as the configuration permits the hydraulic pressure in the communication passages to be properly imposed to the disc valves.

Any number of the respective disc valves may be suitably selected so that the most optimum damping force characteristic may be attained corresponding to the vehicle in question.

It is further possible in the respective embodiments that the extension side damping force generation mechanism provided on a piston and the contraction side damping force generation mechanism provided at the bottom side of the hydraulic damper may be suitably combined and that the damping force characteristic at the extension side may be made different from that of the contraction side or made identical, depending on the requirement.

As explained in detail, the hydraulic damper according to the present invention includes a larger diameter disc valve and a smaller diameter disc valve which are caused to open at different pressure levels are stacked in the order on the downstream side of a communication passage, an inner pressure chamber in communication with the communication passage provided on the side of the larger diameter disc valve facing the communication passage, an outer pressure chamber in communication with the communication passage provided radially outwardly of the outer circumference of the smaller diameter disc valve. As the hydraulic fluid pressure in the outer and inner pressure chambers is increased due to movement of the piston, the outer peripheral portion of the larger diameter disc valve and the entire larger diameter disc valve and the smaller diameter disc valves are sequentially opened so as to generate the

damping force. By changing the rigidity of the disc valves, therefore, the damping force characteristics may be adjusted at will.

In addition, if a pre-set lead setting means is employed, the pressure level at which the smaller diameter disc valve is to open may be changed to a higher level, thus providing a wider range of adjusting the damping force characteristics.

Furthermore, in such a construction as a spring member is employed in place of the smaller diameter disc valve, the damping force characteristics may be freely adjusted by changing the elasticity of a spring member.

In this manner, a most optimum damping force characteristics may be established for a specific vehicle, whereby driving stability and driving comfort are considerably improved.

Furthermore, the disc valve or spring member may be simply mounted and no special technique is needed to assemble them, assembly of the hydraulic damper is very efficient.

We claim:

1. A hydraulic damper comprising:

a cylinder,

a piston slidably fit in said cylinder and partitioning the interior of said cylinder to two liquid chambers, said piston being provided with a communication passage formed therethrough for communicating said two liquid chambers with each other, and

a damping force generation mechanism for generating damping force by controlling the flow of liquid through said communication passage caused by sliding movement of said piston in the cylinder,

wherein said damping force generation mechanism includes:

a larger diameter disc valve disposed on the downstream side of said communication passage,

a smaller diameter disc valve stacked on said larger diameter disc valve, the diameter of said smaller diameter disc valve being smaller than that of the larger diameter disc valve,

an inner pressure chamber disposed radially inwardly on the opposite side of said larger diameter disc valve remote from the smaller diameter disc valve, said inner pressure chamber being in communication with said communication passage, and

an outer pressure chamber disposed on the same side as said inner pressure chamber radially outwardly of said inner pressure chamber, said outer pressure chamber being

located radially outwardly of the outer circumference of said smaller diameter disc valve, said outer pressure chamber being in communicate with said communication passage.

2. A hydraulic damper as claimed in Claim 1, wherein said damping force generation mechanism further includes a pre-set load setting means for giving said smaller diameter disc valve an initial deflection.

3. A hydraulic damper as claimed in Claim 1, wherein said smaller diameter disc valve is adapted to provide a fulcrum for said larger diameter disc valve adjacent to a partitioning wall defined between the inner and outer pressure chambers when the outer circumference portion of said larger diameter disc valve is deflected.

4. A hydraulic damper as claimed in Claim 3, wherein said partitioning wall is provided with a throttling passage, said inner and outer pressure passage being in communication with each other through said throttling passage.

5. A hydraulic damper as claimed in Claim 1, wherein said damping force generation mechanism further includes a partitioning wall defined between said inner and outer pressure chambers, said partitioning wall being provided with a throttling passage formed therein for communicating said inner and outer pressure chambers with each other.

6. A hydraulic damper as claimed in Claim 1, wherein said larger diameter disc valve is provided with a plurality of holes formed in an area thereof which is overlaid with

said smaller diameter disc valve, said holes being located adjacent the outer circumference of said smaller diameter disc valve.

7. A hydraulic damper comprising:

a cylinder,

a piston slidably fit in said cylinder and partitioning the interior of said cylinder to two liquid chambers, said piston being provided with a communication passage formed therethrough for communicating said two liquid chambers with each other, and a damping force generation mechanism for generating damping force by controlling the flow of liquid through said communication passage caused by the sliding movement of said piston in the cylinder,

wherein said damping force generation mechanism includes:

a disc valve disposed on the downstream side of said communication passage,

inner and outer pressure chambers disposed at different locations in the radial direction of said disc valve and respectively being in communication with said communication passage, said inner and outer pressure chambers being adapted to impose hydraulic pressure on said disc valve, and

a biasing means disposed on the opposite side of said disc valve remote from said pressure chambers for biasing said disc valve toward said pressure chambers, said biasing means being located in the location corresponding to that of a partition wall separating said inner and outer pressure chambers.

8. A hydraulic damper substantially as described herein with reference to and as illustrated in Figures 3 and 4 of the accompanying drawings.

9. A hydraulic damper substantially as described herein with reference to and as illustrated in Figures 5 and 6 of the accompanying drawings.

10. A hydraulic damper substantially as described herein with reference to and as illustrated in Figure 7 of the accompanying drawings.

11. A hydraulic damper substantially as described herein with reference to and as illustrated in Figures 8 and 15 of the accompanying drawings.

12. A hydraulic damper substantially as described herein with reference to and as illustrated in Figures 9 and 15 of the accompanying drawings.

13. A hydraulic damper substantially as described herein with reference to and as illustrated in Figure 10 of the accompanying drawings.

14. A hydraulic damper substantially as described herein with reference to and as illustrated in Figure 11 of the accompanying drawings.

15. A hydraulic damper substantially as described herein with reference to and as illustrated in Figure 12 of the accompanying drawings.

16. A hydraulic damper substantially as described herein with reference to and as illustrated in Figure 13 of the accompanying drawings.

17. A hydraulic damper substantially as described herein with reference to and as illustrated in Figure 14 of the accompanying drawings.

18. A hydraulic damper substantially as described herein with reference to and as illustrated in Figures 15 and 18 of the accompanying drawings.

19. A hydraulic damper substantially as described herein with reference to and as illustrated in Figures 16 and 18 of the accompanying drawings.

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